EXPERIMENTAL DETERMINATION OF TRANSVERSE TEMPERATURE PROFILES FOR OIL FLOWING IN A CIRCULAR TUBE

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PREFACE

The measurement of local temperatures in temperature gradient fields of high Prandtl Number fluids at high pressures does not appear to be adequately described in the literature. The need for this data to verify assumptions made in theoretical analysis in heat transfer as well as to use in calculating bulk temperatures has prompted this investigation.

I wish to thank Dr. J. H. Boggs for granting me a graduate assistantship from Oklahoma State University. My deepest thanks go to Dr. J. D. Parker and Dr. J. A. Wiebelt for their counsel throughout the course of this work as my advisers.

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I owe a debt of gratitude to my wife, Ann, for her patience and help during the course of my study. Finally, I especially acknowledge my parents for their support and guidance without which I would not have obtained a college education.

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CHAPTER I

INTRODUCTION

One of the most common industrial methods for changing the temperature of a fluid is to pass it through a tube which has a surrounding temperature that is different than the fluid. This difference in temperature causes an energy exchange between the tube and the fluid thereby changing the fluid temperature and creating a transverse local temperature profile as well as an axial one. This same principle was used in this investigation for cooling oil that was flowing inside a horizontal tube by forced convection. The oil had been heated to a temperature greater than the surroundings by viscous shear dissipation and energy addition by the pump.

The most common experimental means of determining the transverse local temperature profile of a flowing fluid is to measure it with a micrometric traversing temperature probe device that disturbs the original conditions of the fluid by an insignificant amount. The apparatus devised, technique used, and the results obtained for the fluid moving at rates up to approximately 13 feet per second are presented in this paper.

McAdams (1) states that this experimentally determined transverse temperature profile information about liquids flowing inside a tube is of considerable interest, both in revealing the mechanisms by which the heat is transferred between wall and fluid and for confirming assumptions made in deriving theoretical relations that involve rates of heat transfer between the tube and the fluid. Theoretical heat transfer analysis in this area (See Selected Bibliography) contains assumptions about transverse local temperature profiles which need to be verified by test.

On the other hand, there are three approaches to the experimental determination of the so-called mixing-cup or bulk temperature of a specified mass of oil flowing past a given cross section of a tube. The first is to measure the fluid temperature resulting from the perfect mixing in a "cup" of the fluid flowing past a certain cross sectional area of the tube per unit time. This method requires a mixing chamber and is not applicable in a case such as this when two successive measurements are to be made at different stations with only a few inches between them along the tube.

The second approach is to consider the use of a system of fast acting valves and a bypass line in order to "trap" a segment of the fluid between two of these valves. An arrangement of thermocouples can be used to measure temperatures at various points between the two closed valves. Use can be made of these temperatures to calculate the bulk temperature. This method also was discarded because it was too clumsy and uneconomical.

The third approach was the one considered the most practical in view of the facts presented in the two previous paragraphs. The third approach was used in this investigation. Knudsen and Katz (2) present an equation that enables one to calculate the bulk temperature from known temperature and velocity profiles. This thesis does not include the calculation of bulk temperatures, but shows how they may be calculated.

Sheppard (3) and Woolfenden (4) are two men who have measured local transverse temperature profiles of liquids flowing in a circular tube. Pertinent parts of their papers that relate to this investigation are discussed in Chapter II of this thesis.

This investigation differs from the two mentioned above in that the tube size is smaller and the 850 pounds per square inch pressure is much higher. In addition to the differences mentioned above, a traversing thermistor probe device was used instead of a traversing thermocouple probe. The merits of the thermistor in lieu of the thermocouple for sensing temperature differentials are discussed in Chapter III of this thesis.

In summarizing it may be stated that the objective of this investigation was to design, build, and test a traversing thermistor probe mechanism which would be capable of measuring local temperatures and sensing small temperature differences without leaking oil when used in pressure fields of a maximum pressure of 2,000 pounds per square inch and of a maximum temperature of 200°F.

CHAPTER II

PREVIOUS INVESTIGATIONS OF TEMPERATURE PROFILES

Sheppard (3) built and tested an apparatus for making micrometric temperature traverses across fluids moving at rates up to 15 feet per second in a 1 1/2-inch brass tube. The tube was surrounded by a 4 1/2-inch steam jacket for heating the hydrocarbon oil in one case and water in the other. Vertical flow was used to minimize asymmetrical free convection effects.

The traversing probe consisted of a brass tube of 1/8-inch outside diameter. Glass insulated iron-constantan 24-gage thermocouple leads were drawn through the center of the tube and sealed in it with a kaolin-sodium silicate cement.

The two 24-gage thermocouple leads protruded 1/4 of an inch from the junction end of the probe and were formed into a pronged fork 1/8 of an inch across points. Sheppard found that 24-gage wire was the smallest that could be used to provide adequate stiffness to the probe tip.

Several junctions were calibrated in a constant temperature oil bath against standard thermometers and were found not to vary appreciably from the Leeds and Northrup calibration for standard iron-constantan wire. Therefore, Leeds and Northrup millivolt versus temperature tables were used to convert potential difference readings to their corresponding temperatures.

To position the thermocouple, Sheppard used a 6-inch vernier depth gauge with a least count of 0.001 inch. For traversing near the wall for the oil flowing in the tube, a dial indicator with a least count of 0.0001 inch was used.

Traverses were made in the test oil at Reynolds Numbers between 7,000 and 14,000 and with Prandtl Numbers between 63 and 206. For a typical example, one run showed that for external steam heating and a mean Reynolds Number of 10,350, the temperature of the inside tube wall was 225°F. while the temperature at the centerline of the tube was 173.5°F. Over 90% of this temperature difference occurred within 1% of the distance from the tube wall to the axis of the tube.

Sheppard's conclusions were that his experimental technique yielded data sufficiently accurate for the study of fluid flow versus heat phenomena.

Woolfenden (4) built and tested a traversing temperature probe device to measure local transverse temperature profiles for water flowing in an externally heated 2-inch diameter copper tube. A steam jacket was used for the external heating.

A pipe length of 100 diameters upstream from the temperature measuring station was used to insure smoothing of all irregularities introduced into the fluid stream by components located upstream.

Two mean temperatures were distinguished as the mean cross sectional temperature and the mean fluid temperature. The mean cross sectional temperature is an average for an area and is not to be confused with the mean fluid temperature; i. e., bulk temperature. The bulk temperature is the average temperature for a given volume of fluid and is of value in making energy balances.

To determine the mean fluid temperature or bulk temperature it is helpful to consider the pipe cross section divided into annular elements dA. If the temperature of each annular element is given the symbol t_f, then the bulk or mixing cup temperature may be expressed as

$$t_b = \frac{2\pi \int_0^R t_f \, u \, r \, dr}{2\pi \int_0^R u \, r \, dr} \tag{2-1}$$

In order to calculate t_b, the transverse temperature and velocity profiles must be known as a function of r.

To measure the temperature profile Woolfenden used thermocouples made of No. 30 copper and ideal wire. Rubber insulation was applied to the lead wires and they were threaded through a 1/16-inch copper tubing which was used for the probe.

The sliding fit between the tube wall and the probe stem did not seal the water in the tube which was subjected to 40 pounds per square inch pressure. The leakage was said to be small enough to neglect.

The copper tube used for the probe was bent to a 90° angle about 3 inches from the hot junction. The portion of the probe away from the hot junction was passed through a 3/8-inch diameter brass pipe and sealed inside of it with melted sulfur.

To locate the hot junction, the thermocouple wires were connected across a buzzer and battery to the copper pipe. The hot junction was then moved until the buzzer sounded which indicated that the junction was touching the inside pipe wall.

Leakage around the hole in the pipe through which the thermocouple probe entered was allowed as an escape for the air liberated from the water by the heating. It seems that this caused Woofenden to obtain non-symmetrical profiles. However, the difference in water density at different temperatures may have been a contributing cause also.

Temperature distribution was taken from top to bottom and then back. The velocity ranged from 51.3 to 523 feet per minute. The pipe temperature ranged from 57.5 to 95°C. while the bulk temperature varied from 24.8 to 50°C. He recommended vertical pipe position instead of horizontal to remove the effect of gravity and the difference in condensate film thickness. Woofenden says runs at higher velocity and smaller pipe diameter would be of much interest.

A typical run at an average velocity of 523 feet per minute inside the 2-inch pipe produced temperature data that made an almost symmetrical profile when plotted as temperature versus radial location in the pipe. At the centerline of the pipe the measured temperature was 21°C. while at the inside pipe wall the temperature was about 30°C. The bulk temperature for this flow condition was 24.8°C.

CHAPTER III

DESIGN OF TRAVERSING DEVICE

The oil whose temperature was to be measured flowed inside of a smooth, horizontal, stainless steel tube which had an inside diameter of 0.622 inches and an outside diameter of 0.750 inches. The mechanical details of the traversing device which was used are drawn full scale in Figure 1.

A 0.040-inch outside diameter stainless steel tube was used to house the 0.004-inch diameter platinum-iridium lead wires. This was the smallest tube feasible in view of the fact that it had to be bent with a 90° angle and the thermistor leads had to be threaded into a small ceramic tubing which was in turn placed inside the stainless steel tube.

A thermal analysis and calculations were made on the probe to substantiate the fact that stem conduction errors were negligible for the probe designed. This analysis is presented in Chapter IV.

A micrometer depth gage with a least count of 0.001 inch was used to measure the position of the sensing element by accurately known increments between the centerline of the tube and the pipe wall. Plate I shows the depth gage incorporated with the traversing device.

To avoid leakage from 2,000 pounds per square inch system to the atmosphere, O-ring seals were used. The dimensions of the reciprocating

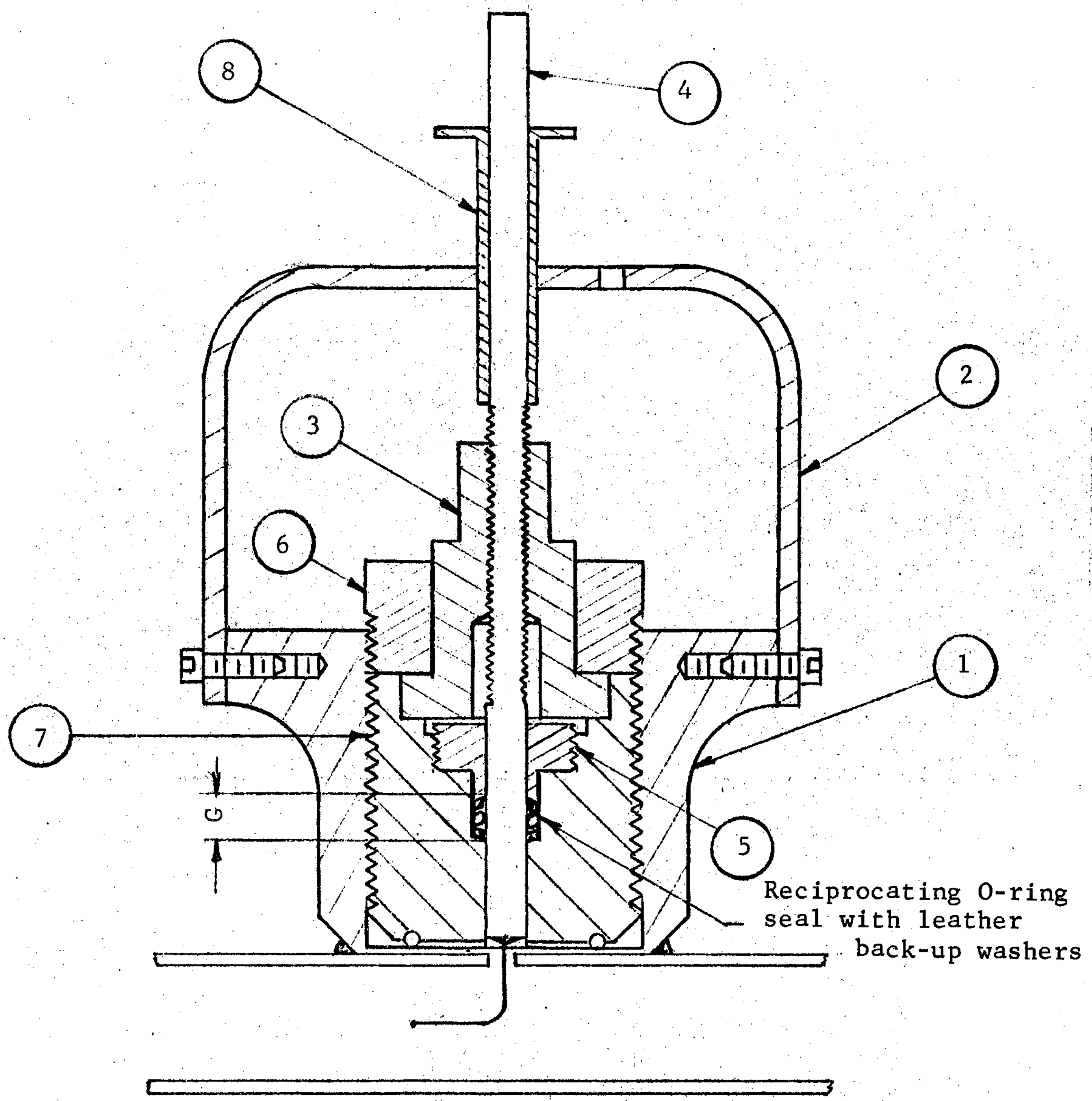
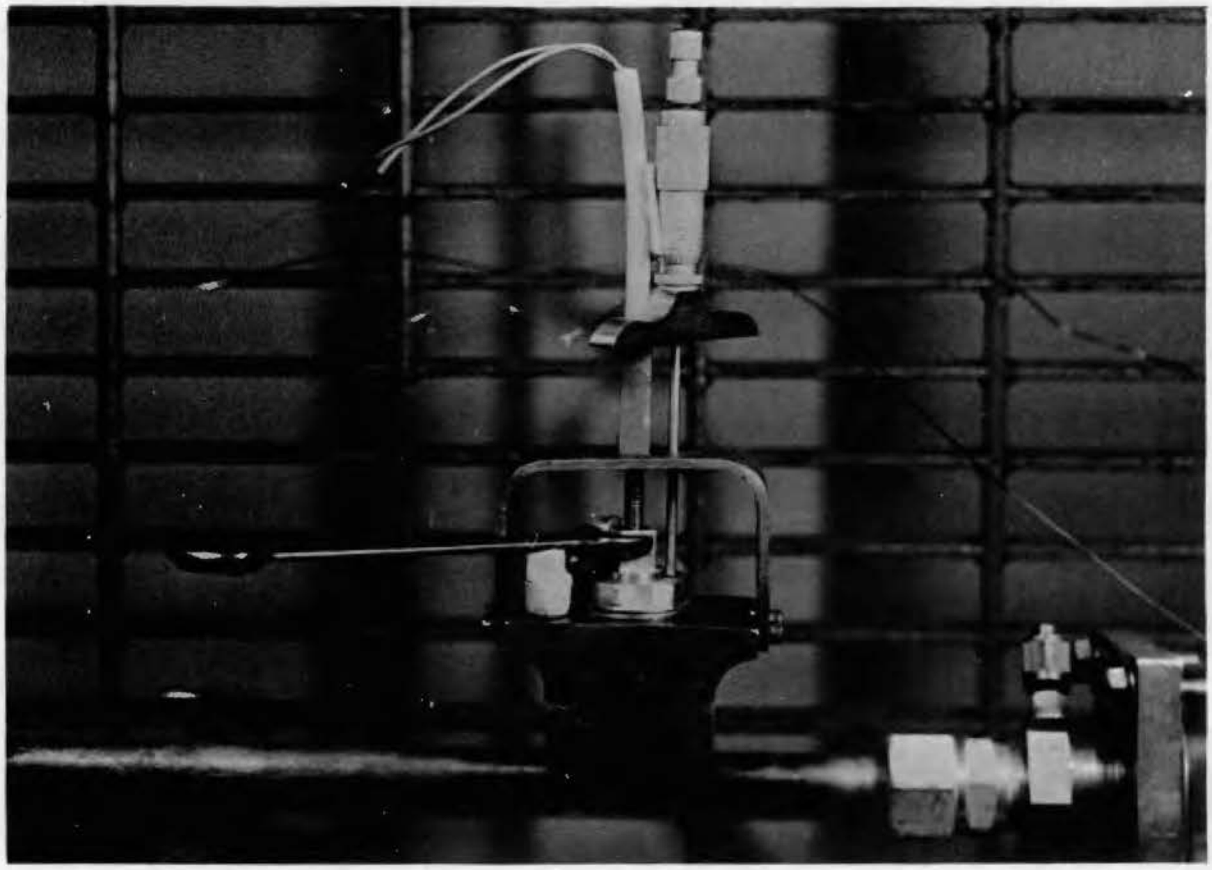


Figure 1. Traversing Mechanism Assembly

Plate I.

Front Vie~ of Test Facility.



fluid-tight seal were taken from Catalog 5700 of the Parker Seal Company of Cleveland, Ohio. The gland is shown in Figure 1.

The temperature traversing mechanism was designed so that the seal would be accessible for replacement of the O-ring or for any of the parts. Close tolerances were used to obtain a fluid tight seal. Note that the assembly of the O-ring onto the piston produced a radial tension and that the cylinder compressed the outside of the O-ring. Leakage is bound to occur when the O-ring is not pre-tensed and compressed in this manner.

An O-ring made of BUNA N, a synthetic rubber, with hardness shore A of 70 was selected for use in this application because BUNA N is more oil resistant, more wear resistant, and retains its flexibility (for the temperature range -40/300°F.) better than any other oil resistant rubber developed. (6).

Leather back-up washers were used to prevent extrusion of the O-ring since pressures greater than 1,500 pounds per square inch were anticipated. They were soaked in oil before installation to make them function properly when under pressure. Their function was to prevent extrusion of the O-ring which is the primary cause of reciprocating O-ring seal failure in high pressure application. Extrusion of the O-ring occurs when part of the O-ring material is forced into the clearance between the piston and cylinder and may be actually "bitten off."

The dimension G shown in Figure 1 was obtained by adjusting part number 5. Part number 5 was necessary because of the impossibility of making a boring tool small enough and at the same time rigid enough to machine the O-ring groove into the wall of a single part composed of part number 5 and part number 7.

Finishes of surfaces contacting the O-ring in a moving seal application like this one should be between 5- and 10-micron root mean square for maximum O-ring life. All sharp edges were rounded off to prevent nicking of the O-ring during assembly. Finishes smoother than 5-micron root mean square were avoided because the O-ring tends to remove lubricant from the metal as it moves across the metal surfaces that are too smooth. To assure good lubrication of all metal parts in contact with the O-ring, they were oiled before assembly. After assembly the oil being sealed provided adequate lubrication.

After a general investigation, a thermistor (thermal-sensitive resistor) temperature sensing element was selected. The main reason for choosing a thermistor was that the thermistor had a high negative temperature-coefficient of resistance. This simply means that for a small temperature rise there is an accompanying large decrease of electrical resistance and vice versa. Temperature measurement with thermistors involves the same principle as conventional resistance thermometry. A resistance-measuring circuit was used which contained a 45-volt battery, a wheatstone bridge (see equipment list for description), and the thermistor. According to Shoop and Tuve (7), this temperature-coefficient of resistance of thermistors is about ten times as large as that of metals used in electrical-resistance thermometry. Schiff (8) states that this high temperature-coefficient of thermistors means higher sensitivity than the more common thermocouples or resistance thermometers. This sensitivity provided an excellent means for sensing minute temperature changes that would not have been practical with thermocouples or metal resistance thermometers.

Another advantage of this thermistor was that its terminal resistance within the temperature range of 32°F. to 240°F. was more than 1000 ohms which was enough to make the lead wire resistance negligible. Also, advances in solid-state physics have made it possible to produce thermistors which can maintain their electrical characteristics for years without significant drift even after continuous recycling of temperature over spans of 400°F. (9). This makes it unnecessary to recalibrate.

The small size of 0.043-inch outside diameter of the thermistor bead gave it a fast response time because of its low heat capacity. The time constant was only two seconds. Atkins (10) claims that thermistors are at their best in the medium temperature range (which was the range of interest in the paper) where thermocouples are often inadequate.

From the practical standpoint it was more convenient to use a thermistor than a thermocouple because there was no reference junction, therefore, no reference junction compensation and no polarity checks.

The thermistor selected for use had a code number of GA66J1. It was manufactured by Fenwal Electronics of Framingham, Massachusetts. The bead of this thermistor was composed of mixtures of powdered metallic oxides such as manganese, nickel, cobalt, copper, iron, and uranium. The powdered mixture was sintered into a 0.015-inch diameter bead on two 0.004-inch diameter platinum-iridium lead wires that were tight, parallel, and about 0.010 inches apart. The resulting semiconducting bead was glass coated for protection. Thermistors preaged and sealed in glass beads of this nature have been known to retain their calibration for months within less than 0.02°F. variation, but 1°F. drifts occur if the beads are not sealed in glass. (11). Drifts due to pressure variation do not exceed 0.002°F. per atmosphere. (11). In this case of 850 pounds per square inch maximum or 58 atmospheres, the maximum drift would be 0.116°F. This is less than the accuracy of the calibration so it can be disregarded. The resistance of this thermistor at 77°F. was rated at 6.25 megohms with a tolerance of 20%. The measured resistance at 77°F. was 5.2 megohms.

The resistance of the thermistor is solely a function of its absolute temperature. (9). For sensing the temperature of the oil it was highly important that the amount of power being dissipated within the bead did not change the bead temperature by an appreciable amount; in other words, did not self-heat it.

A value of 0.7 milliwatts which is called the dissipation constant has been furnished for this thermistor. The dissipation constant gives an approximation of the amount of power in milliwatts this glass coated bead will dissipate while raising the thermistor temperature 1°C. above its surroundings. This value was measured at 25°C. with the thermistor suspended by its leads in still air. However, for the case of flowing oil, the dissipation constant may be ten times the value obtained for still air because the thermal conductivity of the oil was higher than air and the forced convection carried away more of the heat dissipated by the thermistor. (12). Therefore, a dissipation constant of 7 milliwatts was allowed for the following analysis.

Because the thermometer was to be used for relatively precise measurements of oil temperature, the design temperature difference allowed was 0.1°F.

$$P_D = \frac{5}{9} D_C \Delta T_s \quad (12) \qquad (3\text{-}1)$$

Substituting 2 milliwatts for D_C and 0.1°F. for ΔT_s into equation (3-1) and solving for P_D gives 0.389 milliwatts.

Power dissipated, potential difference, and resistance for this direct current circuit are related by the expression,

$$R = \frac{E^2}{P_D} \quad . \qquad (3\text{-}2)$$

Substituting the battery voltage of 45 volts and a P_D of 0.389 milliwatts into equation (3-2) and solving gives an R of 6,000 ohms. This resistance corresponds to a temperature greater than encountered. (See Figure 2). This is to say that the 45-volt battery will not cause self-heating of more than 0.1°F. when the resistance is greater than 6,000 ohms. For all the resistance measurements made, the resistance was greater than 6,000 ohms.

However, in order to use the 45-volt source contained in the wheatstone bridge to obtain less than the 0.1°F. self-heating, the battery switch was tapped when making resistance measurements on the bridge. The circuit was closed for only short bursts at ten to fifteen second intervals. By making quick test bursts, excellent results were achieved with the 45 volts supplied to the wheatstone bridge even though some self-heating of the thermistor would have occurred for continuous

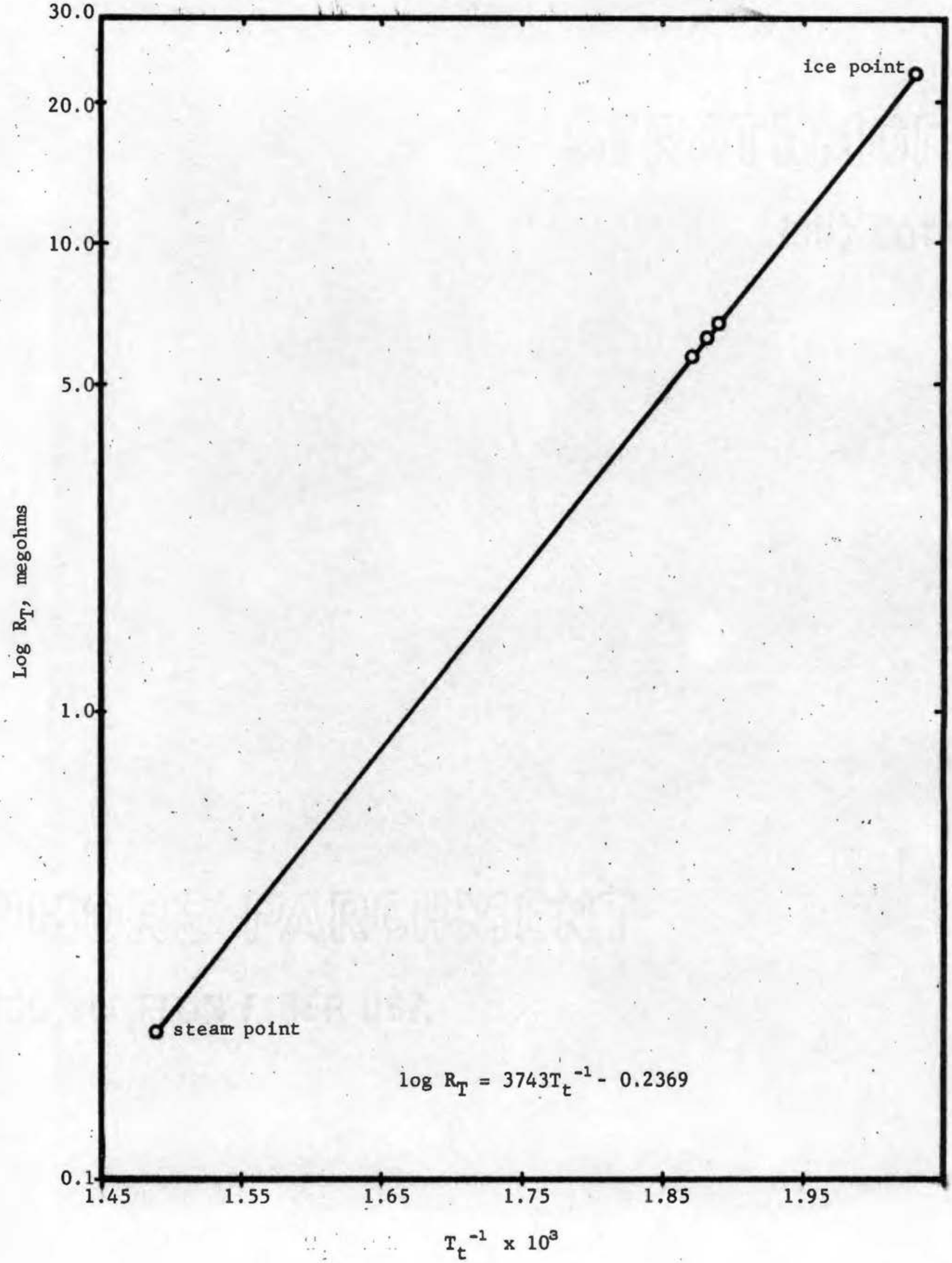


Figure 2. Thermistor Calibration

closure of the battery switch. (13).

The time constant for the thermistor was given as two seconds. This two seconds is the time required in seconds for the thermistor to change its own temperature 63% of the difference between its temperature and the surroundings. If, for example, the thermistor happened to be 1°F. above its surroundings, it would take six seconds for it to return within 0.2°F. of its surroundings.

The design of the probe mechanism included the probability of a self-excited vibration being initiated by vortex formation. When oil flowed around the cylindrical body of the probe, boundary layer separation and vortex formation occurred. (14). Eventually, these vortices broke off and flowed downstream. It was important to know the frequency at which the vortices are shed to see if they would affect the temperature indicated by the thermistor.

Roshko (15) measured the Strouhal Number for flow around a circular cylinder and plotted it on the ordinate against Reynolds Number on the abscissa. For Reynolds Numbers between 800 and 10,000 the Strouhal Number is constant at 0.18.

Schlichting (14) gives the Strouhal Number as

$$S = \frac{n\,D}{V} \tag{3-3}$$

where n = frequency at which vortices are shed in seconds^{-1}

S = Strouhal Number

D = diameter of probe in meters

V = velocity of fluid in meters per second

Calculations in Chapter IV gave, for a flow rate of 12 gallons per minute in the tube used, a mean velocity of 12.7 feet per second or 3.88 meters per second. The diameter of the probe was 0.04 inches or 1.02×10^{-3} meters. Substituting these values of D, S, and V into equation (3-3) and solving yields a frequency of 685 seconds^{-1}.

The time constant for the thermistor bead was 2 seconds. Therefore, the frequency at which the vortices peeled off the probe was too high to affect the temperature measurement.

After all of the parts of the traversing mechanism were made, the assembly began. First, it was necessary to thread both of the 3-inch lengths of 0.004-inch diameter platinum lead wires attached to the thermistor bead into short pieces of ceramic tubing. The pieces of the ceramic tubing had an outside diameter of 0.022 inch with two 0.005-inch holes parallel to the axis through them. Next the ceramic tubing with the thermistor leads inside was threaded into a 1 1/2-inch length of 24-gage hypodermic tubing. The ceramic tubing was used because it was a good electrical insulator as well as a thermal one.

The third step was to drill a 0.0465-inch diameter hole through a piece of 1/4-inch plate steel stock. Then a piece of 1/4-inch diameter steel bar stock was clamped at the edge of the hole. Next, the thermistor bead end of the hypodermic stock was inserted into the 1/4-inch plate until only the bead protruded on the opposite side. At that time, the 90° bend with a 1/8-inch radius was made in the probe by bending the probe around the piece of 1/4-inch steel bar stock. After bending the probe stem, a 1/2-inch length of sleeve with an outside diameter of 0.115 inch and a 0.043-inch diameter hole through it, was threaded onto what was to be the vertical portion of the probe. Then, two 18-inch lengths of platinum lead wire were soldered onto the existing 3-inch leads.

The lead wires were then threaded through a 6-inch length of ceramic tubing. The outside of this piece of 1/8-inch diameter ceramic tubing had to be sanded until its outside diameter was 0.115 inch which allowed it to slip through the inside of part number 4. The sleeve and the piece of bent hypodermic stock were then sealed together with a mixture of one-half epoxy resin and one-half epoxy hardener. The resin and the hardener were contained in separate tubes which are shown in Plate II.

The epoxy was also used to seal the sleeve to part number 4. Epoxy was also placed around the probe tip to keep oil from leaking around the bead into the hypodermic tubing.

In the meantime part number 1 had been silver soldered onto the 3/4-inch outside diameter stainless steel tubing. A 1/8-inch hole was drilled into the pipe wall to allow the thermistor probe stem to be inserted into its operating position. With part number 4 in its operating position, the lead wires were threaded through part number 7 and it was screwed into its position shown in Figure 1 by the large barrel wrench shown on Plate III. A number 211 Parker O-ring was used for a static seal between parts 1 and 7.

Next a back-up washer and then the O-ring and then another back-up washer were threaded onto part number 4 and placed in their operating

Plate II.

View of Hypsometer, Wheatstone Bridge, and Ohmeter.

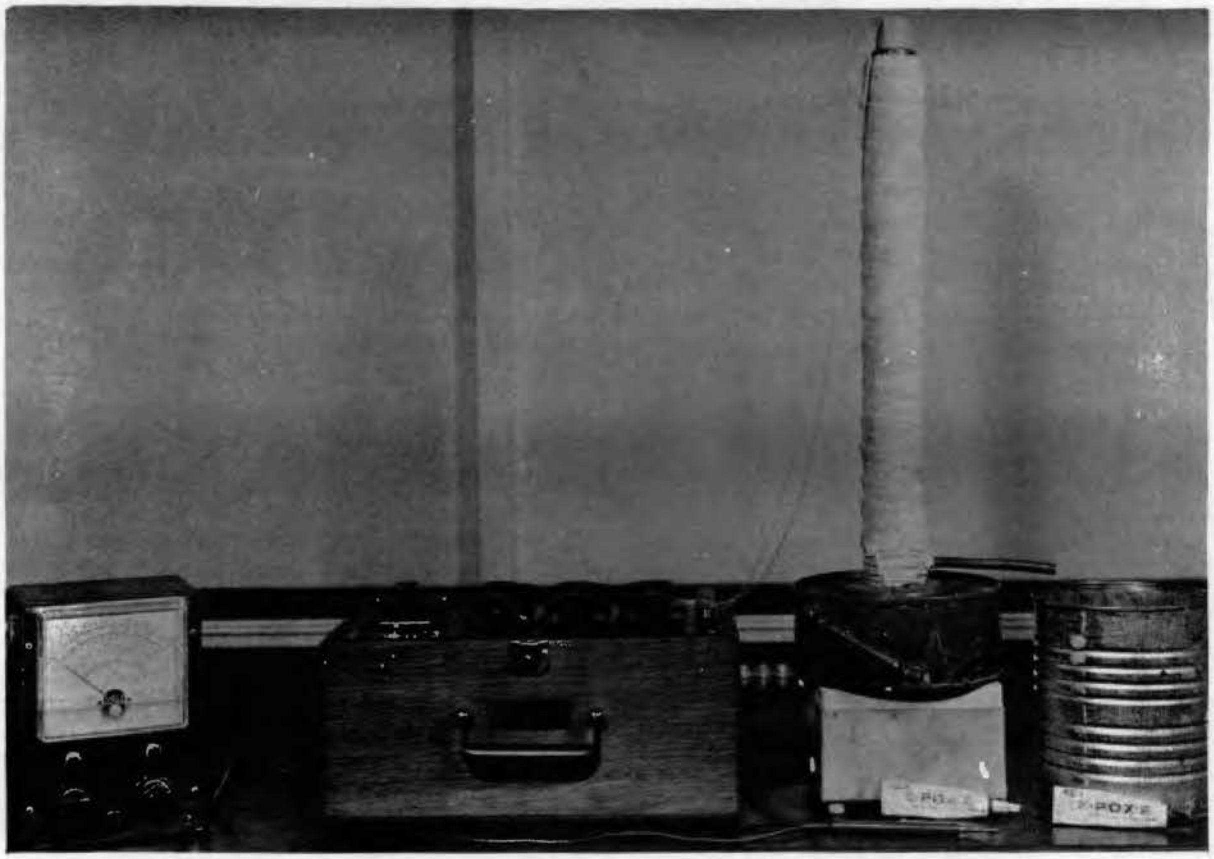
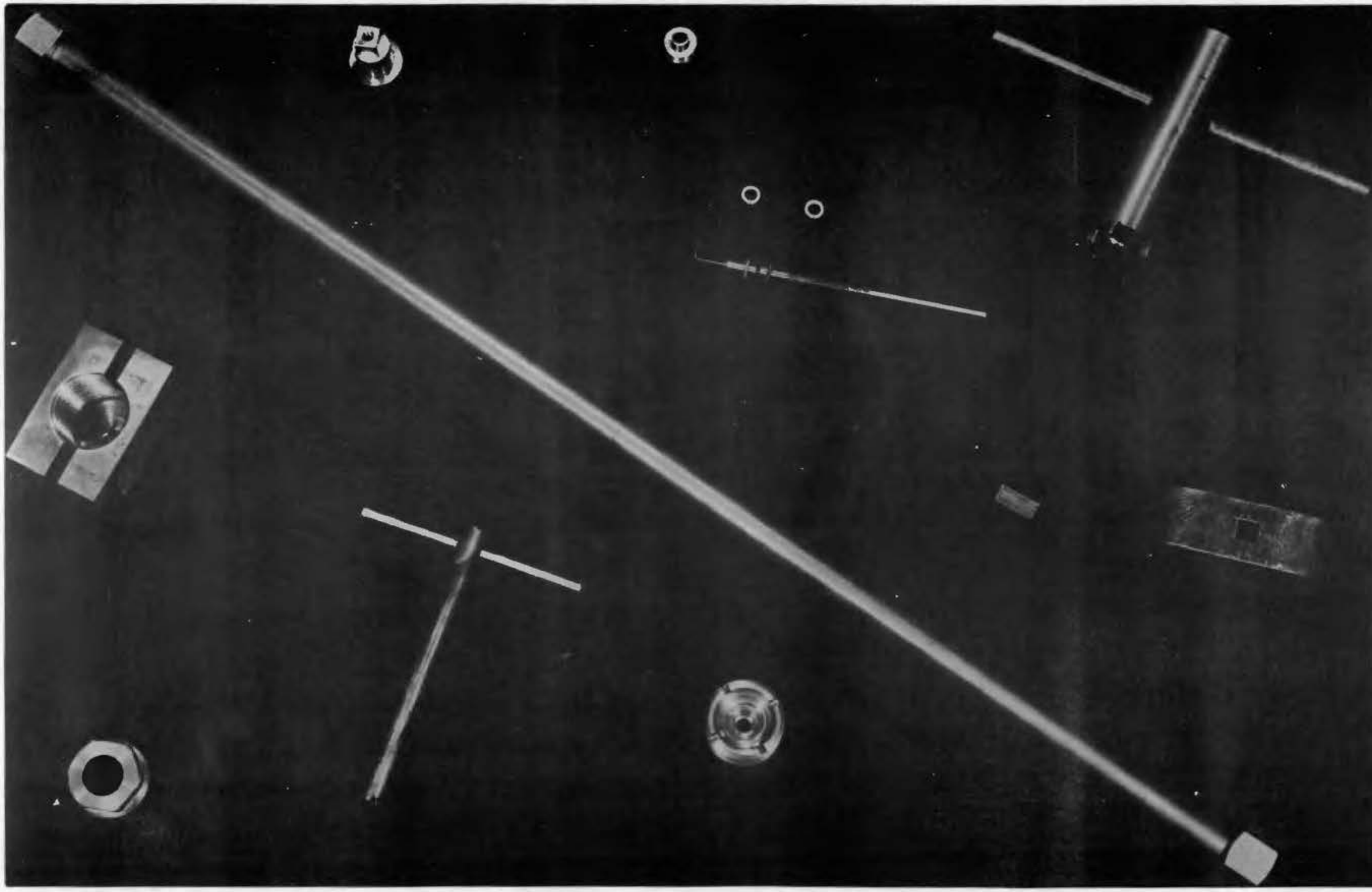


Plate III.

Parts of the Traversing Mechanism.



positions. Part number 5 was screwed in with the small barrel wrench shown in Plate III until it touched the top back-up washer. The function of part number 5 was to hold the back-up washers and O-ring in place, not to press them together.

The next part to be assembled was part number 3. Then part number 6 was screwed into place. This left part number 2 to be screwed into position. Part number 8 was clamped to part number 4 by two headless set screws after having been adjusted to accommodate the depth gage micrometer as shown in Plate I.

After the mechanism was placed into the hydraulic circuit, it was tested. No leakage of the traversing device was detected. The performance of the system as a whole was satisfactory.

CHAPTER IV

ANALYSIS OF PROBE

For sensing local temperatures in a temperature gradient field of flowing fluid it was necessary to design a probe. The main problem was to determine the simultaneous effect that the conduction and convection of heat had on the temperature indicated by a thermistor bead placed on the tip of the probe. A probe that would minimize this effect was built and tested successfully.

Another related problem was to calculate what temperature a thermistor bead would indicate at a certain point in the flowing fluid for a calculated fluid temperature from an assumed local temperature profile of the fluid at the same point.

Nusselt in his conductivity measurements, according to Jakob (16), found that the leads from the sensing element should be led along an isothermal path for some length before leading them across a non-isothermal path. This isothermal length for a particular case was calculated.

My analysis was made subject to the following major assumptions:

1. Radiation heat transfer between the probe and the surroundings was negligible.
2. The system was in a steady state with respect to heat transfer.
3. The transverse fluid temperature gradient was linear.
4. The convective coefficients h_1 and h_2 are constant.

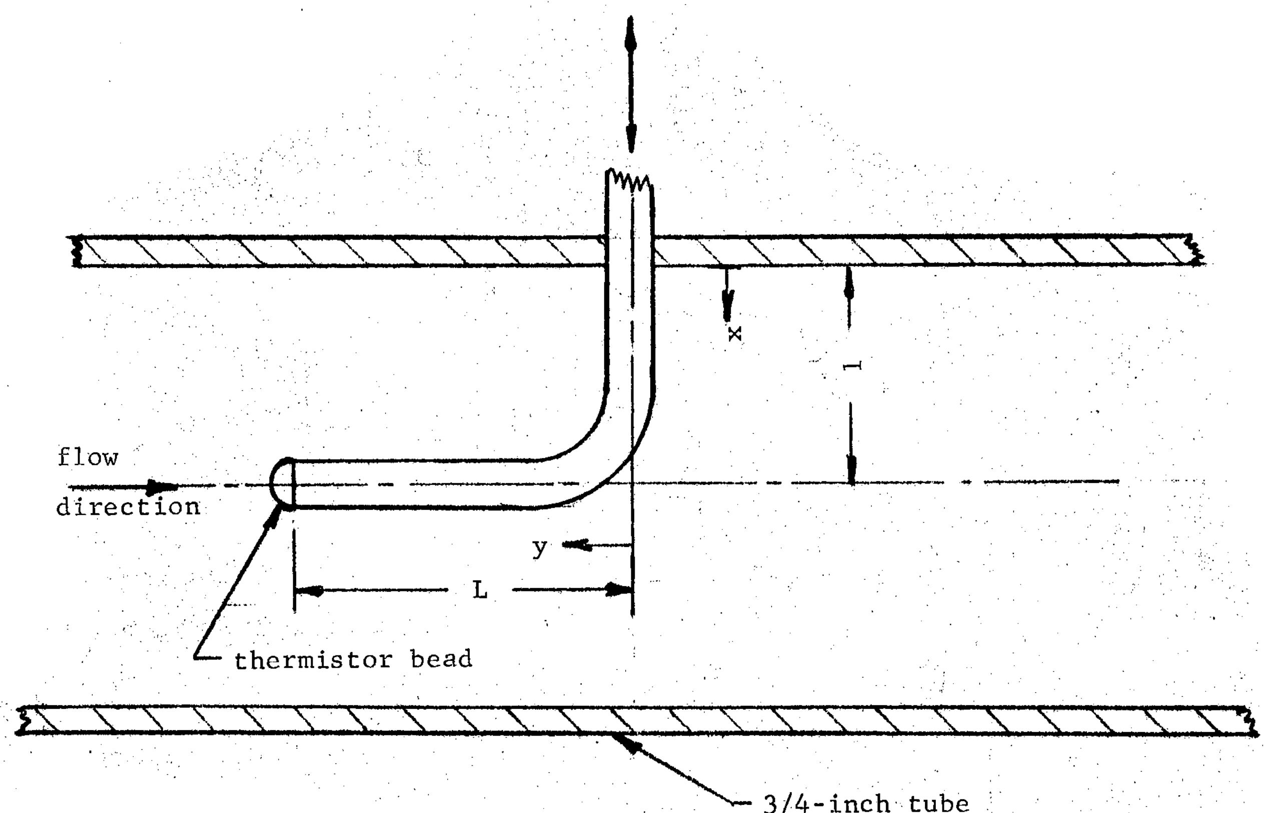


Figure 3. Probe Schematic

To carry out a solution, it was necessary to write two heat balances for differential elements of the probe stem both on the vertical portion of the probe and on the horizontal portion of the probe. By making use of an assumed linear fluid temperature profile and the four boundary conditions listed below, a solution was effected. In particular, the difference between the bead temperature and the fluid temperature was minimized.

The four boundary conditions are:

1. $T\Big|_{x=o} = t_w$

2. $T\Big|_{x=l} = T\Big|_{y=o}$

3. $\frac{dT}{dx}\Big|_{x=l} = \frac{dT}{dy}\Big|_{y=o}$

4. $\frac{dT}{dy}\Big|_{y=L} \cong o$

A heat balance on a differential element of the vertical probe stem yields:

$$q_a = q_b - q_c \tag{4-1}$$

where

$$q_a = - KA \frac{dT}{dx} \tag{4-2}$$

and

$$q_b = -KA \frac{dT}{dx} - KA \left(\frac{d^2T}{dx^2}\right) dx \tag{4-3}$$

$$q_c = h_1(2\pi r_p) \; dx \; (t_f - T) \tag{4-4}$$

Now substituting equations (4-2), (4-3), and (4-4) into (4-1)yields after introducing $\alpha_1^2 = \frac{2h_1}{Kr_p}$, and (4-28)

$$\frac{d^2T}{dx^2} - \alpha_1^2 T = -\alpha_1^2 t_f = -\alpha_1^2 (t_w + cx) \tag{4-5}$$

The complete solution of (4-5) becomes

$$T = A_1 \sinh \alpha_1 x + A_2 \cosh \alpha_1 x + t_w + cx \tag{4-6}$$

When the No. 1 Boundary Condition is applied to equation (4-6) the result is

$$T\Big|_{x=o} = A_2 + t_w \tag{4-7}$$

Since $T\Big|_{x=o}$ is the same as t_w, A_2 is equal to zero.

Now differentiating equation (4-6) with respect to x gives for A_2 equal to zero

$$\frac{dT}{dx} = A_1 \alpha_1 \cosh \alpha_1 x + c \tag{4-8}$$

Evaluating equation (4-8) at x=l gives

$$\frac{dT}{dx}\Big|_{x=l} = A_1 \alpha_1 \cosh \alpha_1 l + c \tag{4-9}$$

A heat balance for a differential element of the horizontal part of the probe stem yields

$$q_d = q_e + q_f \tag{4-10}$$

where
$$q_e = -KA \frac{dT}{dy} \tag{4-11}$$

and

$$q_d = - KA \frac{dT}{dy} - KA \left(\frac{d^2T}{dy^2}\right) dy \qquad (4-12)$$

and

$$q_f = h_2 \; (2\pi r_p) dy \quad (t_f - T) \qquad (4-13)$$

Substituting equations (4-11), (4-12), and (4-13) into (4-10) produces after inserting $\alpha_2{}^2 = \frac{2h_2}{Kr_p}$, and (4-28)

$$\frac{d^2T}{dy^2} - \alpha_2{}^2 \; T = - \alpha_2{}^2 \; t_f = - \alpha_2{}^2 \; (t_w + cl) \qquad (4-14)$$

The complete solution of equation (4-14) may be written as

$$T = A_3 \; \sinh \; \alpha_2 \, y + A_4 \; \cosh \; \alpha_2 \, y + t_w + cl \qquad (4-15)$$

Now with No. 2 Boundary Condition, equation (4-15) becomes

$$T\Big|_{x=l} = T\Big|_{y=o} = A_4 + t_w + cl \qquad (4-16)$$

Evaluating equation (4-6) with $A_2 = o$ at $x = l$ gives

$$T\Big|_{x=l} = A_1 \; \sinh \; \alpha_1 \, l + t_w + cl \qquad (4-17)$$

Solving equation (4-16) and (4-17) simultaneously gives

$$A_4 = A_1 \; \sinh \; \alpha_1 \, l \qquad (4-18)$$

The evaluation of equation (4-8) at $x = l$ gives

$$\frac{dT}{dx}\Big|_{x=l} = A_1 \, \alpha_1 \; \cosh \; \alpha_1 \, l + c \qquad (4-19)$$

Differentiating equation (4-15) with respect to y yields

$$\frac{dT}{dy} = A_3 \alpha_2 \cosh \alpha_2 y + A_4 \alpha_2 \sinh \alpha_2 y \tag{4-20}$$

Evaluating equation (4-20) at y equal to zero produces

$$\left.\frac{dT}{dy}\right|_{y=o} = A_3 \alpha_2 \tag{4-21}$$

Equation (4-21) and (4-9) and Boundary Condition No. 3 combined result as

$$A_3 \alpha_2 = A_1 \alpha_1 \cosh \alpha_1 l + c \tag{4-22}$$

Boundary condition No. 4 and equation (4-20) combined give

$$A_3 \alpha_2 \cosh \alpha_2 L + A_4 \alpha_2 \sinh \alpha_2 L = o \tag{4-23}$$

Solving equation (4-18) and (4-22) simultaneously eliminates A_1 and yields

$$A_3 \alpha_2 = \frac{A_4 \alpha_1 \cosh \alpha_1 l}{\sinh \alpha_1 l} + c \tag{4-24}$$

Noting that equation (4-23) and (4-24) contain A_3 and A_4 algebraic manipulation gives

$$A_3 = \frac{c \tanh \alpha_1 l \tanh \alpha_2 L}{\alpha_1 + \alpha_2 \tanh \alpha_1 l \tanh \alpha_2 L} \tag{4-25}$$

and

$$A_4 = \frac{-c \tanh \alpha_1 l}{\alpha_1 + \alpha_2 \tanh \alpha_1 l \tanh \alpha_2 L} \tag{4-26}$$

Evaluating equation (4-15) at y = L yields

$$\left. T \right|_{y=L} = A_3 \sinh \alpha_2 L + A_4 \cosh \alpha_2 L + t_w + cl \tag{4-27}$$

To calculate the temperature of the probe tip under these circumstances from equation (4-27) it was necessary to obtain values for the parameters.

When the horizontal portion of the probe was on the centerline of the tube the value of l was 0.0259 ft.

The assumed linear fluid temperature profile mentioned previously may be expressed as

$$t_f = t_w + cx \tag{4-28}$$

where c is a positive constant. The value of t_f equals t_w when x is zero at the tube wall. However, at the centerline of the tube where x equals l the fluid temperature was assumed to be 5° F. greater than the fluid temperature at the tube wall. This fixed the value of c at 192° F. per foot.

For MIL-H-5606A, the hydraulic oil used, the values of following properties at 200°F. were obtained. (17).

$$\rho_f = 1.67 \text{ slugs/ft}^3$$

$$\mu_f = 0.211 \text{ slugs/ft-hr}$$

$$C_p = 16.75 \frac{\text{Btu}}{\text{slug-}^\circ\text{F.}}$$

$$K_f = 0.076 \text{ Btu - hr}^{-1}\text{- ft}^{-1} \text{ - }^\circ\text{F.}^{-1}$$

The relation for the Prandtl Number is

$$N_{Pr} = \frac{\mu_f \, C_p}{K_f} \tag{4-29}$$

Substituting values into (4-29) gives a value of 46.5 for N_{Pr}.

The cross sectional flow area for the 0.622-inch inside diameter tube may be expressed as

$$A_t = \frac{\pi(D_i)^2}{(4)\ 144} \qquad (4\text{-}30)$$

Substituting 0.622 inch for D_i into equation (4-30) and solving yields an A_t of 0.00211 ft^2.

For a flow rate of 12 gallons per minute or 0.0267 ft^3/sec through the tube, the mean velocity was calculated from

$$V_m = \frac{Q}{A_t} \qquad (4\text{-}31)$$

and found to be 12.68 feet per second.

The calculation of the convective coefficient h_2, on the horizontal portion of the probe was made by means of equation (4-33), an equation established by Sieder and Tate (18).

The Reynolds Number used in equation (4-33) may be written as

$$N_{Re} = \frac{V_m\ L\ \rho_f}{\mu_f} \qquad (4\text{-}32)$$

The length L, the dimension of the horizontal part of the probe, was made to be 1/2 of an inch. This was the greatest length that could have been used for the method of assembly discussed in Chapter III. The N_{Re} was calculated from equation (4-32) to be 1.51×10^4.

According to Sieder and Tate (18),

$$N_{Nu} = 0.027\ (N_{Re})^{0.8}\ (N_{Pr})^{0.33} \left(\frac{\mu_f}{\mu_{fw}}\right)^{0.14} \qquad (4\text{-}33)$$

All of the fluid properties are evaluated at the bulk temperature except μ_{fw} which is evaluated at the wall temperature. The last term in equation (4-33) was calculated to be 0.986. Substituting the appropriate values into equation (4-33) and solving yields a value of 213 for N_{Nu}.

Since the Nusselt Number is written as

$$N_{Nu} = \frac{h_2 L}{K_f} \quad , \qquad (4\text{-}34)$$

the value of h_2 was calculated to be 389 Btu-hr^{-1}-ft^{-2}-$^{\circ}$F^{-1} after substituting the proper values into equation (4-34) and solving.

Jakob and Hawkins (16) present the following relation for correlating the variables which influence the heat transfer coefficient, h_1, for oil flowing normal to the vertical portion of the probe.

$$N_{Nu} = 0.6\ (N_{Re})^{0.5}\ (N_{Pr})^{0.31} \qquad (4\text{-}35)$$

The characteristic length used for calculating the Reynolds Number was the probe diameter. The Reynolds Number for equation (4-35) was calculated to be 1190. Solving equation (4-35) for the N_{Nu} gives 74.6. From

$$N_{Nu} = \frac{h_1\ D_p}{K_f} \qquad (4\text{-}36)$$

h_1 was determined to be 1700 Btu-hr^{-1}-ft^{-2}-$^{\circ}$F.$^{-1}$.

The value of the thermal conductivity for stainless steel was taken from Jakob and Hawkins (16) as K = 9.5 Btu-hr^{-1}-ft^{-2}- F.$^{-1}$.

The values of α_1 and α_2 were calculated with an equivalent radius of 0.0143 inch or 11.9 x 10^{-4} feet and found to be 173 ft^{-1} and 252 ft^{-1} respectively. The equivalent radius was the radius of a hypothetical solid probe that had the same cross sectional area as the hypodermic tubing which had an outside radius of 0.02 and an inside radius of 0.014 inch.

The values of A_3 and A_4 were obtained by equations (4-24) and (4-25). With equation (4-27) the temperature of the probe tip was calculated to be the same as the fluid temperature at the same point on the axis of the tube. In other words the error was extremely small.

The same approach was used with L equal to zero, that is to say a probe without a horizontal segment. The result was that the difference between the calculated probe tip temperature and the fluid temperature at the location of the probe tip was 0.4°F.

For a L of 0.3 inches an error of approximately 0.1°F. was obtained. In conclusion, it may be stated that the L of 0.5 inches was conservative and that the design was satisfactory.

CHAPTER V

EXPERIMENTAL PROCEDURE AND RESULTS OF TEST

The stagnation temperature of the fluid in contact with the probe tip was calculated to be 0.005°F. greater than the local fluid temperature immediately upstream from the probe tip. This difference was neglected since it was in existence to some extent while all of the data was being taken.

Upstream from the probe tip a distance of 24 inches, the oil flowed into the test pipe section from a high pressure rubber hose. For laminar flow a value of 7 inches was calculated as an adequate starting length for this case. (5). Chapman (5) states that no adequate theory exists which will predict the starting length in turbulent flow since the nature of the pipe entrance, pipe roughness, and other factors have a serious effect on the growth and transition of the boundary layer in the pipe. The use of 24-inch unobstructed pipe length upstream from the probe was considered satisfactory to insure smoothing of all irregularities introduced into the fluid by upstream components.

At a pressure of 1500 pounds per square inch the traversing mechanism was tested. No leakage occurred during the time that the traverse was being made nor before or after it was made.

It was necessary to wait between one and two hours after starting the hydraulic circuit to obtain essentially steady state conditions. Measurements indicated that 0.1°F. variations in a local temperature occurred in a fifteen minute interval even after the system oil temperature gage had been constant for fifteen minutes beforehand. For this reason traverses were made after this variation had decreased to an unmeasurable value with the instrumentation used.

Traverses were made in one direction and then the opposite direction with no appreciable differences after the transient heat transfer period was over. Some measurements were made that were not reproducible and had to be rejected.

In order to position the thermistor bead accurately, a depth gage micrometer of a least count of 0.001 inch was used. Before the test pipe section was assembled into the hydraulic circuit, a micrometer reading was recorded with the thermistor bead touching the pipe wall. This enabled one to know the location of the probe tip and to adjust it to a desired position by turning the wrench shown in Plate I.

The first four runs used were taken with the probe tip being traversed vertically and the last four runs with the probe tip being traversed horizontally. This was accomplished by rotating the pipe 90°. It was not possible to detect significant differences in the shape of the temperature profiles plotted from the data due to the orientation of the traversing mechanism. This information leads to the fact that the temperature distribution was symmetrical to the axis of the tube.

By changing the flow rate of the cooling water to the heat exchanger, the reservoir temperature was adjusted to predetermined values. In all cases the temperature gradient near the center of the tube was measured to be zero.

An attempt was made to find what effect the flow rate had on the temperature profile. The approach was to maintain, as nearly as possible, a constant system pressure and reservoir temperature while varying the flow rate.

Other tests were made to find what effect the viscosity had on the temperature profile. To do this the system pressure and flow rate were held as constant as possible while the reservoir temperature was changed.

The data and calculated results are tabulated in Tables I and II. The local temperatures were calculated from the measured thermistor resistance by the use of equation (8) in Appendix C.

TABLE I

DATA AND CALCULATED RESULTS FOR MIL-H-5606A

Distance from Tube Wall	Run 1 R_T	Run 1 t_f	Run 2 R_T	Run 2 t_f	Run 3 R_T	Run 3 t_f	Run 4 R_T	Run 4 t_f
10^{-3} inches	Megohms	°F.	Megohms	°F.	Megohms	°F.	Megohms	°F.
Tube Wall 0	3.010	97.7	2.018	112.5	1.236	131.7	0.7165	154.7
15	3.005	97.8	2.012	112.6	1.232	131.85	0.712	155.0
35	3.000	97.8	2.010	112.6	1.230	131.9	0.709	155.2
55	3.000	97.8	2.010	112.6	1.229	131.9	0.709	155.2
105	3.000	97.8	2.010	112.6	1.229	131.9	0.709	155.2
180	3.000	97.8	2.010	112.6	1.229	131.9	0.709	155.2
℄ Tube 311	3.000	97.8	2.010	112.6	1.229	131.9	0.709	155.2
Ø of system	500		500		500		500	
Q of system	7.85		7.85		7.95		7.90	

TABLE II

DATA AND CALCULATED RESULTS FOR MIL-H-5606A

Distance from Tube Wall	Run 5		Run 6		Run 7		Run 8	
	R_T	t_f	R_T	t_f	R_T	t_f	R_T	t_f
10^{-3} inches	Megohms	°F.	Megohms	°F.	Megohms	°F.	Megohms	°F.
Tube Wall 0	0.691	156.3	0.755	152.4	1.243	131.6	1.338	128.5
15	0.686	156.7	0.751	152.7	1.233	131.8	1.332	128.7
35	0.682	156.9	0.747	152.9	1.225	132.0	1.331	128.7
55	0.679	157.1	0.747	152.9	1.225	132.0	1.331	128.7
105	0.676	157.3	0.747	152.9	1.225	132.0	1.331	128.7
180	0.674	157.4	0.746	152.9	1.225	132.0	1.331	128.7
℄ Tube 311	0.674	157.4	0.746	152.9	1.225	132.0	1.331	128.7
Ø of system	850		500		500		850	
Q of system	11.5		8.0		5.45		11.45	

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

Experience with this equipment showed that it was a balanced design with respect to accuracy. In other words, although it would be relatively easy to extend the accuracy of traversing to a usable least count of 0.0001 inches and temperature measurements to 0.001°F., more accurate results would also require more accurately known fluid properties, more nearly steady state operation, and more precise methods of indicating the tube wall with a smaller diameter thermistor bead. A prime requisite would be a recording galvanometer and other sensitive instrumentation that would plot temperature variation in the turbulent flow so that time average temperatures could be easily and accurately recorded at each traverse point.

It is recommended that two thermistors instead of one be used in the future. The placing of the two thermistors in different locations, one at a known temperature and one in the temperature gradient field, will make the unbalance on the wheatstone bridge dependent upon the difference of temperature of the two thermistors. This method has been used on alternating current with a high gain amplifier on the output of the bridge so that temperature differentials of 0.001°F. could be measured easily.

In addition to the present problem, the equipment might be adapted to the study of temperature profiles resulting from a combination of free and forced convection.

In the tests run the tube wall was uninsulated. However, in future tests it may become desirable to know the inside tube wall temperature for steady state heat transfer conditions. Since it is inconvenient to measure this temperature, calculations were made to show that the measurement of the outside wall temperature would give essentially the same temperature as the inside tube wall temperature under certain conditions. These conditions are that a 1-inch thick layer of glass wool be placed around the circumference of the tube when operating between 100°F. and 200°F. If the wall is adiabatic there will be no temperature gradient in fluid except due to viscous heating.

A circulating oil bath with a good temperature control device and a standard platinum resistance control thermometer would make it possible to calibrate the thermistor or thermistors much more accurately.

The data for Runs 1 and 2 was not plotted because there was obviously nothing to be gained by plotting this information. The rest of the runs were plotted on Figures 4 and 5. From Figure 4 it may be seen that an increase in flow rate will decrease the difference in the temperature of the turbulent core and the fluid temperature at the tube wall.

The fact that the slope of the curve plotted for Run 7 is greater than the one plotted for Run 3 would indicate that more heat was conducted through the boundary layer for laminar flow conditions of Run 7 than for the turbulent flow conditions of Run 3.

Figure 5 shows the reproducible nature of the temperature profiles. From Table II, the data from Runs 5 and 8 were plotted. The slope of the curve plotted for Run 5 was the greater. This information may be interpreted that the lower viscosity of the oil in Run 5 resulted in more heat conduction through the boundary layer. On the other hand, the greater difference between the fluid temperature at the wall and at the axis of the tube, indicates greater radial heat transport in the turbulent core for Run 5. This was caused by the higher temperature and the lower viscosity, but the ratio of their individual contributions was not determined.

In evaluating the suitability of the bead thermistor for sensing the local temperature profiles, it was found that a smaller diameter bead on the order of 0.010 of an inch or smaller would have been better than the 0.043-inch diameter bead used.

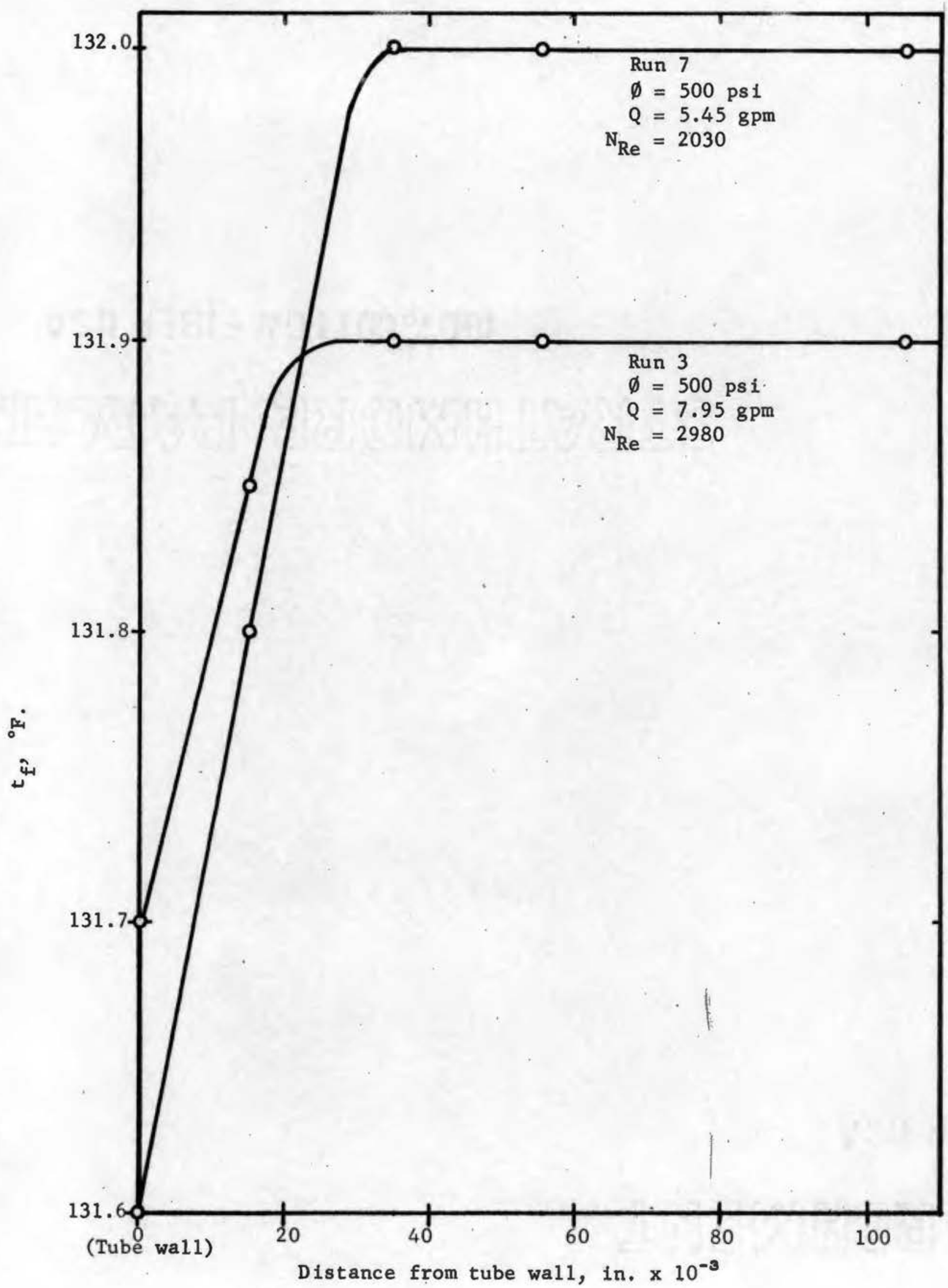


Figure 4. Graphical Interpretation of Data

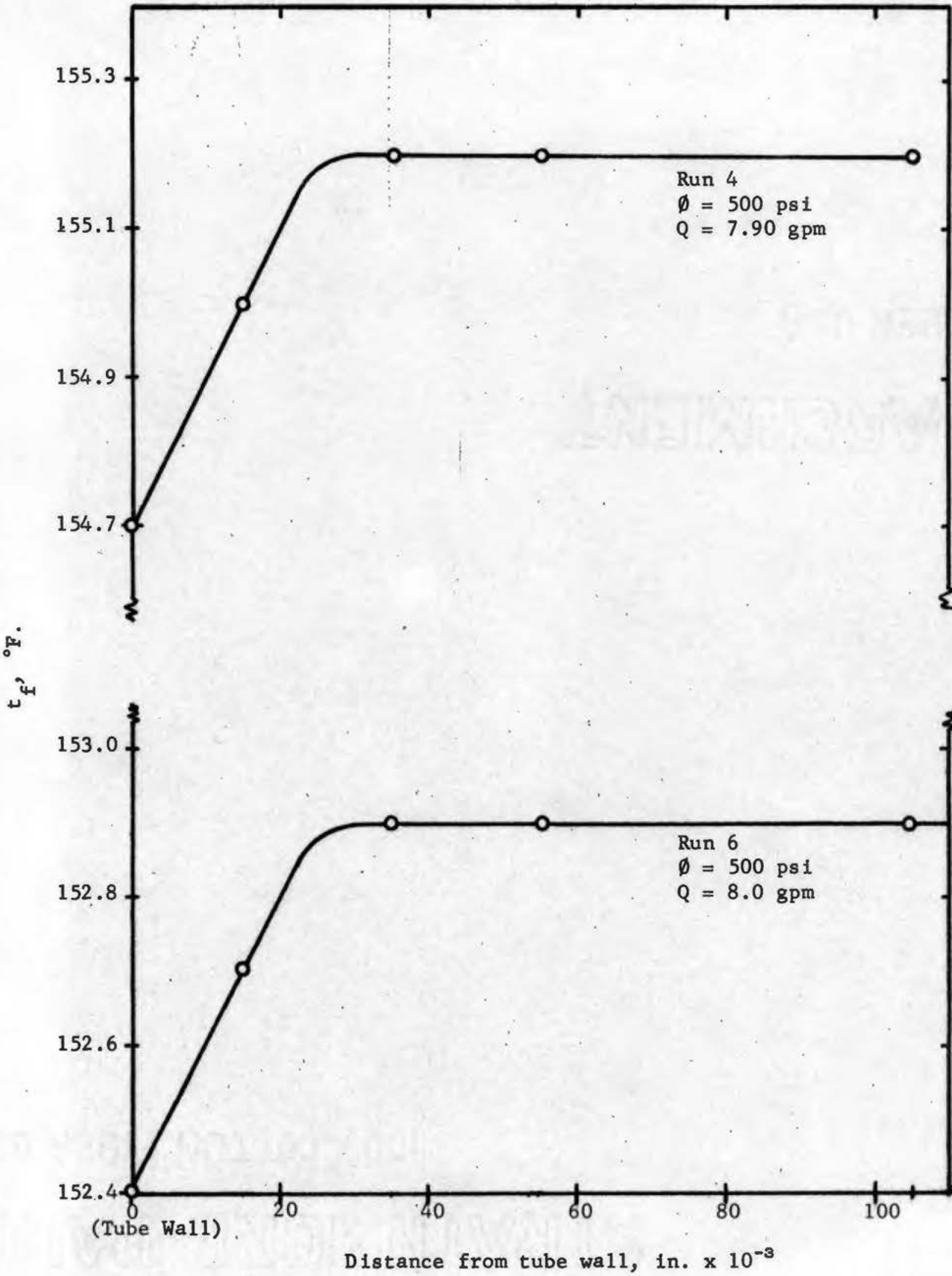


Figure 5. Graphical Interpretation of Data

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APPENDIX A

NOMENCLATURE

A	cross sectional area of probe stem, square feet
A_1	a constant, degrees Fahrenheit
A_2	a constant, degrees Fahrenheit
A_3	a constant, degrees Fahrenheit
A_4	a constant, degrees Fahrenheit
a	a constant, dimensionless
A_t	flow area of tube, square feet
c_p	specific heat of fluid, Btu/slug - degrees Fahrenheit
c	a positive constant, degrees Fahrenheit/foot
D_c	dissipation constant of thermistor bead, milliwatts/ degrees Centigrade
D	diameter of probe stem, meters
d	prefix, indicating derivative, dimensionless
D_i	inside diameter of pipe, feet
D_p	diameter of probe stem, feet
E	battery voltage, volts
h_1	convection coefficient of heat transfer on vertical part of probe, Btu/(hr)(square feet)(degrees Fahrenheit)
h_2	convection coefficient of heat transfer on horizontal part of probe, Btu/(hr)(square feet)(degrees Fahrenheit)
K_f	thermal conductivity of fluid, Btu/(hr)(feet)(degrees Fahrenheit)

K thermal conductivity of stainless steel probe, Btu/(hr)(feet)($^{\circ}$F.)

l length of vertical portion of the probe, feet

L length of horizontal portion of the probe, feet

n frequency at which vortices are shed, seconds^{-1}

N_{Pr} Prandtl Number of fluid, dimensionless

N_{Re} Reynolds Number, dimensionless

N_{Nu} Nusselt Number, dimensionless

P_D power dissipated in the thermistor bead, milliwatts

Q fluid flow rate, cubic feet per second

q_a steady state heat conduction into differential element on the vertical portion of the probe stem, Btu/hour

q_b steady state heat conduction away from differential element on the vertical portion of the probe stem, Btu/hour

q_c steady state heat transfer into differential element on the vertical portion of the probe stem by convection, Btu/hour

q_d steady state heat conduction away from differential element on the horizontal portion of the probe stem, Btu/hour

q_e steady state heat conduction into differential element on the horizontal portion of the probe stem, Btu/hour

q_f steady state heat transfer into differential element on the horizontal portion of the probe stem by convection, Btu/hour

R total inside tube radius, feet

r local tube radius, feet, measured from tube axis

r_p radius of probe stem, feet

R_T thermistor bead resistance, ohms

R_1 resistance of standard carbon resistor, ohms

R_{eq} equivalent resistance of R_1 and R_T in parallel

S Strouhal Number, dimensionless

t_b bulk temperature of fluid, degrees Fahrenheit

t_f	local temperature of fluid, degrees Fahrenheit
t_w	temperature of inside tube wall, degrees Fahrenheit
T	local temperature of the probe stem, degrees Fahrenheit
T_t	absolute temperature of thermistor, degrees Rankine
u	local velocity of fluid, feet per second
V	mean velocity of fluid, meters per second
V_m	mean velocity of fluid, feet per second
x	vertical coordinate along the probe at right angle to y-axis, feet
y	horizontal coordinate along the horizontal portion of the probe, feet

Greek Letters

α_1^2	parameter identified as $\frac{2h_1}{Kr_p}$, feet^{-2}
α_2^2	parameter identified as $\frac{2h_2}{Kr_p}$, feet^{-2}
β	a constant, degrees Rankine
ΔT_s	design temperature difference allowed between the thermistor bead and the immediate surroundings, degrees Fahrenheit
π	pi, a constant of 3.1416
μ_f	local absolute viscosity of fluid, slugs/foot-hour
μ_{fw}	absolute viscosity of fluid at the inside tube wall, slugs/foot-hour
ρ_f	density of fluid, slugs/cubic foot
$\emptyset$	oil pressure, pounds per square inch

Abbreviations

sinh	hyperbolic sine
cosh	hyperbolic cosine
tanh	hyperbolic tangent
MIL	military
H	hydraulic
ft	feet
hr	hour
$^{\circ}$F.	degrees Fahrenheit
$^{\circ}$C.	degrees Centigrade
$^{\circ}$R.	degrees Rankine

Symbols

%	percent
℄	centerline of tube

APPENDIX B

APPARATUS AND EQUIPMENT

1. Precision Wheatstone Bridge: Manufacturer, Minneapolis-Honeywell Regulator Co.; Model No. 412A.

2. Bead Thermistor: Manufacturer, Fenwal Electronics; Serial No. GA66J1.

3. Laboratory Hypsometer: Manufacturer, Oklahoma State University Mechanical Engineering Laboratory.

4. O-Rings: Manufacturer, Parker Seal Co.; Sizes No. 10 and 211.

5. Hydraulic Test Circuit: Manufacturer, Oklahoma State University Mechanical Engineering Laboratory.

6. Volt Ohmyst: Manufacturer, Radio Corporation of America; Serial No. 35760.

7. Micrometer Depth Gage: Manufacturer, Starrett Co.; No. 445.

APPENDIX C

THERMISTOR CALIBRATION

Before starting calibration a new 45-volt battery was installed in the wheatstone bridge. This was to insure maximum sensitivity of the galvanometer.

The steam data point was taken by using a laboratory hypsometer. The probe was immersed into a long glass tube filled with oil. Steam was condensed on the outside of the tube for almost four hours before the resistance of the thermistor was measured. The steam point at the time of the resistance measurement was 210.4°F. This value was obtained from the Keenan and Keyes Steam Tables (19) for saturated steam under a barometric pressure of 29.005 inches of mercury. The thermistor resistance was 209,740 ohms within an accuracy of ± 20 ohms.

The following table lists other data points that were taken and ascertained to be reproducible. This data in Table III was taken with the probe immersed in a small oil bath with a mercury in glass thermometer. The change in bath temperature was caused by change in room temperature at a rate of approximately 1°F. per hour. Some data points were taken and discarded because they were not reproducible. The values of resistance in the table below are accurate to three significant figures. This is all that the sensitivity of the galvanometer would allow.

TABLE III

Temperature (°F.)	Resistance (ohms)
69.5	6.77×10^6
71.5	6.40×10^6
72.5	6.215×10^6
76	5.65×10^6

The determination of the ice point resistance of the thermistor bead required that a standard resistor be placed in parallel with the bead. This was necessary since the resistance of the thermistor at the ice point was greater than the 10 megohms that could be measured on the calibrating bridge.

The thermistor probe was immersed in a glass tube filled with oil and allowed to stand in a thermos bottle filled with crushed ice for nine hours before any measurements were made.

The resistance of the standard resistor in parallel with the immersed thermistor was then measured and found to be 3.755×10^6 ohms. The resistance of the standard carbon resistor was 4.495×10^6 ohms at 76°F. The equivalent resistance of these two resistors in parallel may be expressed as

$$R_{eq} = \frac{R_1 R_T}{R_1 + R_T} \tag{1}$$

where R_T = thermistor resistance at the ice point in ohms

$R_{eq} = 3.755 \times 10^6$ ohms

$R_1 = 4.495 \times 10^6$ ohms

Substitution of the values of R_{eq} and R_1 into equation (1) yields a 22.8×10^6 ohms for R_T at the ice point. This value was checked by an ohmmeter. The equation of the straight line shown on semi-log coordinates in Figure 2 may be expressed as:

$$\log R_T = \beta/T_t + a \qquad (2)$$

where β and a are constants. Therefore, to evaluate the constants for the straight line approximation of the relationship between the thermistor resistance and the reciprocal of the absolute fluid temperature, proceed as follows.

Pick two known points on the line, one near each end of the line to obtain

$R_T = 22.80 \times 10^6$ ohms at $1/T_t = 2.034 \times 10^{-3}$ $1/{}^\circ R.$ and

$R_T = 209{,}740$ ohms at $1/T_t = 1.492 \times 10^{-3}$ $1/{}^\circ R.$

Then, $$\log(22.80 \times 10^6) = 2.034 \times 10^{-3}\beta + a \qquad (3)$$

and $$\log(0.20974 \times 10^6) = 1.492 \times 10^{-3}\beta + a \qquad (4)$$

Now subtracting equation (4) from equation (3) eliminates a and gives

$$\log(22.80 \times 10^6) - \log(0.20974 \times 10^6) = \beta(2.034 - 1.492)\times 10^{-3} \qquad (5)$$

or

$$\log \frac{22.80}{0.20974} = \beta(0.544 \times 10^{-3})$$

Then, $$\beta = \frac{\log 108.7}{0.5440 \times 10^{-3}} = \frac{2.036}{0.5440 \times 10^{-3}}$$

Therefore, the slope β is equal to 3743.

Now noting that equation (2) can be written in the form

$$a = \log R_T - \beta/T_t \tag{6}$$

the following equation is obtained.

$$a = \log R_T - 3742/T_t$$

Now, the constant a may be evaluated by using the values of a known point on the line as follows:

$$a = \log(209.74 \times 10^3) - 3743/670.12 \tag{7}$$

Solving equation 7 for a yields -0.2369.

With the substitution of the values of a and β into equation (2) it becomes

$$\log R_T = 3743\ T_t^{-1} - 0.2369 \tag{8}$$

Equation (8) may now be used to calculate the absolute temperature from the measured thermistor resistance.

VITA

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Candidate for the Degree of

Master of Science

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Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Buffalo, Oklahoma, May 6, 1938, the son of C. Paul and Ara S. Hughes.

Education: Attended grade school in Clinton, Oklahoma; graduated from Elk City High School, Oklahoma in 1956; received the Bachelor of Science degree from Oklahoma State University, Stillwater, Oklahoma, with a major in Mechanical Engineering, May, 1961; completed the requirements for the Master of Science degree in August, 1962.

Experience: Employed by Paul Hughes, Consulting Engineer, from June-September, 1956, as a surveyor's helper; employed by Shell Oil Company from June-September, 1957, as a plant maintenance assistant; employed by United States Department of Agriculture from June-September, 1958, as an assistant on locating flood prevention dams; employed by United States Department of Agriculture from June-September, 1959, as an assistant on designing irrigation systems; employed by Kockum's Mekaniska Verkstads, Malmo, Sweden, from June-September 1960, as an exchange engineer trainee, sponsored by the International Association for the Exchange of Students for Technical Experience; employed by Chevrolet Engineering from June-September, 1961, as a junior engineer; employed by Oklahoma State University from January-May, 1960, and September, 1961-June, 1962, as a graduate assistant.

Honors and Awards: Dean's Honor Roll, Pi Tau Sigma, honorary mechanical engineering, and Sigma Tau, honorary engineering.